1994

EER and Refrigerating Capacity of Dual Split Type Air-Conditioner

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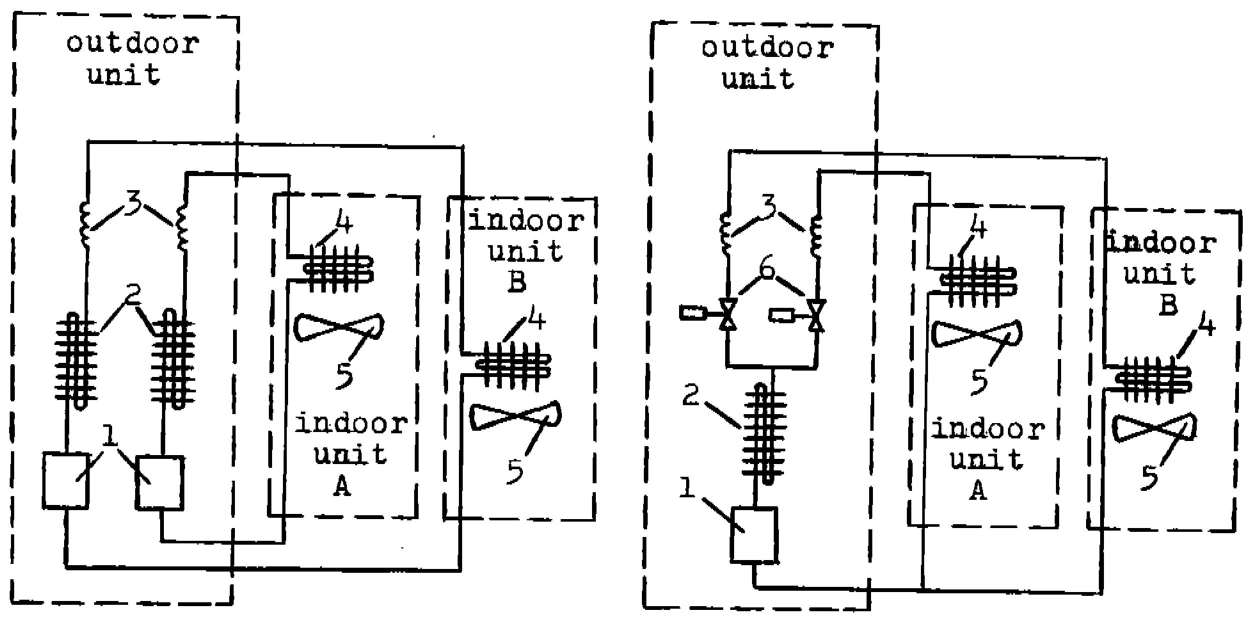
ABSTRACT

There are two kinds of dual split type air-conditioner. One is of using two compressors for two indoor units; the other has only one compressor for two indoor units. The EER and refrigerating capacity of the former kind are just the same as the single split type's. But to one compressor dual split type system, there are two working ways: 1. compressor and both two indoor units operating, and 2. compressor and only one indoor unit operating. In these two ways the EER and refrigerating capacity of the system are different. In this paper, the EER and refrigerating capacity of one compressor dual split type air-conditioner are discussed.

INTRODUCTION

All kinds of household air-conditioner are common nowadsys,



(a). Two compressor system (b). One compressor system

1 — compressor, 2 — condenser, 3 — capillary, 4 — evaporator, 5 — fan, 6 — solenoid valve.

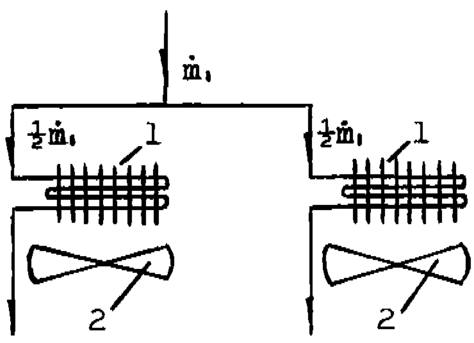
Fig.1 Dual Split Type Air-conditioner System especially window type and single split type. But these two types of air-conditioner can't satisfy all people's needs, so some other kinds of air-conditioner are developed, such as dual split type or even multe split type. In our study, we mainly pay attantion to the dual split type air-conditioner, which has two indoor units installed in two separate rooms. Each indoor unit offers cooling air for one room.

Dual split type air-conditioner has two kinds of system: two compressors dual split system and one compressor dual split system as shown in Fig.1. In fact, the system shown in Fig.1(a) is just the same as two single split type air-conditioners. When a room needs cooling, only one compressor and one indoor unit start to work. When two rooms need cooling, two compressors and two indoor units begin to run. So the EER and refrigerating capacity of two compressors dual split system are like the single split type's. To one compressor dual split system shown in Fig. 1(b), since compressor keeps running in two cases (case 1: two indoor units refrigerating; case 2: only one indoor unit refrigerating), hence, the EER and refrigerating capacity in case 1 are different from the EER and refrigerating capacity in case 2. In this paper, all discussions are based on one compressor dual split system.

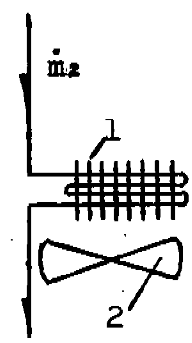
EER AND REFRIGERATING CAPACITY ANALYSES

1. The lgp-h Diagrame of One Compressor Dual Split System

Fig.2 is a schem of two cases of one compressor dual split system operating. Case 1 is two indoor units refrigerating and case 2 is only one indoor unit running.



(a). Case 1: two indoor units refrigerating



(b). Case 2: one indoor unit refrigerating

1 — evaporator, 2 — fan.

Fig.2 Two Cases of One Compressor Dual Split System Operating

Fig.3 is the lgp-h diagrame for both the two indoor units refrigerating and the only one indoor unit refrigerating. In fact, circulation $a_1-b_1-c_1-d_1-a_1$ stands for case 1 and circulation $a_2-b_2-c_2-d_2-a_2$ stands for case 2. It is shown in Fig.3 that evaporating pressure P_{02} and temperature t_{02} of system in case 2 is lower than evaporating pressure P_{01} and temperature t_{01} in case 1. The reason is that when one indoor unit refrigerates, refrigerant flows through only one capillary and one evaporator and the flow resistance increases. This results in the decreas of evaporating pressure and temperature.

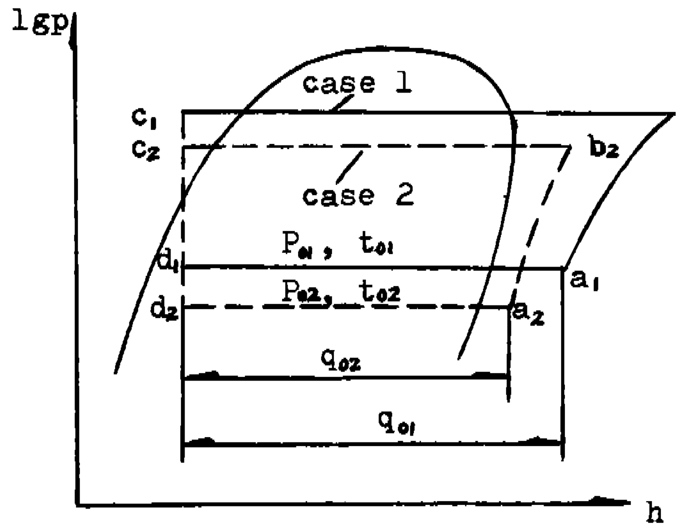


Fig.3 The lgp-h Diagrame of One Compressor Dual Split System

2. Refrigerant Mass Flow Rate $\dot{m}$

The equation of state of real gas is as follows:

$$PV=ZmRT$$ [1]

where, m — mass of gas,
P — pressure of gas,
V — volume of gas,
Z — compressibility factor of gas,
R — gas constant,
T — temperature of gas.

From the equation of state, the relationship between mass flow rate $\dot{m}$ and volume flow rate $\dot{V}$ is

$$P\dot{V}=Z\dot{m}RT$$

Compared with two indoor units refrigerating, when only one indoor unit runs, the volume flow rate $\dot{V}$ of refrigerant gas sucked in by compressor and the compressibility factor Z almost unchange, but the pressure and temperature of gas decreas. Since Kelvin temperature scale is used in the equation of state, the change of temperature affects $\dot{m}$ little. Therefore, main affection to $\dot{m}$ comes from pressure. As we know, the pressure of refrigerant gas sucked in by compressor in case 1 is higher than in case 2, hence, in case 1, refrigerant gas flow rate $\dot{m}_1$ is larger than $\dot{m}_2$ in case 2, i.e. $\dot{m}_1 > \dot{m}_2$.

3. The Mass Performance Factor q_o

In case 1, the mass performance factor q_{o1} is

$$q_{o1} = h_{a1} - h_{d1}$$

where, h_{a1} — enthalpy of refrigerant in point a_1,
h_{d1} — enthalpy of refrigerant in point d_1.

In case 2, the mass performance factor q_{o2} is

$$q_{o2} = h_{a2} - h_{d2}$$

where, h_{a2} — enthalpy of refrigerant in point a_2,
h_{d2} — enthalpy of refrigerant in point d_2.

Since the total air flow rate at two indoor units running is as twice as the air flow rate at one indoor unit running, so the enthalpy value of refrigerant out of evaporator in case 1 is higher than in case 2, i.e. $h_{a1} > h_{a2}$. Therefore,

$$q_{o1} = h_{a1} - h_{d1} > q_{o2} = h_{a2} - h_{d2}$$

4. Refrigerating Capacity $\dot{Q}_o$

Refrigerating capacity $\dot{Q}_o$ can be figured out with nass performance factor q_o and refrigerant mass flow rate $\dot{m}$,

$$\dot{Q}_o = \dot{m} \cdot q_o \quad [2]$$

In case 1, $\dot{Q}_{o1} = \dot{m}_1 \cdot q_{o1}$

In case 2, $\dot{Q}_{o2} = \dot{m}_2 \cdot q_{o2}$

From above discussions, it is known that

$$\dot{m}_1 > \dot{m}_2, \quad q_{o1} > q_{o2}$$

hence, $\dot{Q}_{o1} > \dot{Q}_{o2}$

5. The Indicated Power of Compressor $\dot{W}$

The indicated power consumed by compressor is

$$\dot{W} = \dot{m} \cdot (h_b - h_a)$$

where, h_a — enthalpy in the inlet of compressor,
h_b — enthalpy in the outlet of compressor.

In case 1, $\dot{W}_1 = \dot{m}_1 \cdot (h_{b1} - h_{a1})$

In case 2, $\dot{W}_2 = \dot{m}_2 \cdot (h_{b2} - h_{a2})$

Because the difference between $(h_{b1}-h_{a1})$ and $(h_{b2}-h_{a2})$ is very small, the change of the indicated power comes from the change of refrigerant mass flow rate $\dot{m}$. Since $\dot{m}_1 > \dot{m}_2$, the indicated power of compressor in case 1 is larger than in case 2.

6. EER Value

Due to the very small change of fan power in whole system, the EER value of one compressor dual split system can be expressed approximately as follows:

In case 1, $EER_1 = \dfrac{\dot{Q}_{o1}}{\dot{W}_1}$

In case 2, $EER_2 = \dfrac{\dot{Q}_{o2}}{\dot{W}_2}$

Here, $\dot{Q}_{o1} > \dot{Q}_{o2}, \dot{W}_1 > \dot{W}_2$. Since the difference between $\dot{Q}_{o1}$ and $\dot{Q}_{o2}$ is larger than the difference between $\dot{W}_1$ and $\dot{W}_2$, therefore,

$$EER_1 > EER_2$$

THE TEST RESULTS

The system has been tested according to Chinese National Standard of "Room Air-conditioner GB7725-87". In the test, two indoor units are installed in the same room. First, both of the two indoor units refrigerate, and then, only one indoor unit refrigerates. In these two working conditions, refrigerating capacity of each indoor unit is measured separately. The input power of the system is measured too at the same time. The test results are shown in table 1 and 2.

Table 1. Test results when both two indoor units keep running

test item	t_1 °C	t'_1 °C	t_2 °C	refrigerating capacity W	voltage V	current A	input power W	EER W/W
indoor unit A	27.0	19.4	14.9	1788	—	—	—	—
indoor unit B	26.9	19.3	14.4	1808	—	—	—	—
total	—	—	—	3596	220	6.7	1420	2.53

t_1 — dry bulb temperature of air into the evaporator,
t'_1 — wet bulb temperature of air into the evaporator,
t_2 — dry bulb temperature of air out of the evaporator.

Table 2. Test results when only one indoor unit keeps running

test item	t_1 °C	t_1' °C	t_2 °C	refrigerating capacity W	voltage V	current A	input power W	EER W/W
indoor unit A	26.9	19.3	10.5	2778	220	6.0	1280	2.17
indoor unit B	26.9	19.3	11.0	2808	220	6.0	1280	2.19

t_1 — dry bulb temperature of air into the evaporator,
t_1' — wet bulb temperature of air into the evaporator,
t_2 — dry bulb temperature of air out of the evaporator.

Comparing two indoor units refrigerating with only one indoor unit running, it is known from table 1 and 2 that in case 1 the refrigerating capacity and input power of the system are larger than in case 2, but the change of refrigerating capacity is much larger than the change of input power, hence, in case 1 the EER value of the system is larger than in case 2.

The agreement between test results and theoretic analyses is satisfactory.

CONCLUSION

The one compressor dual split type air-conditioner operates in two ways: two indoor units refrigerating (the first way) and only one indoor unit refrigerating (the second way). The state and mass flow rate of the refrigerant circulating in system are different in these two ways. It has been found that

1. In two indoor units refrigerating, evaporating pressure and temperature are higher than in only one indoor unit running.
2. In two indoor units refrigerating, refrigerant mass flow rate is larger than in only one indoor unit running.
3. In two indoor units refrigerating, refrigerating capacity of the system is larger than in only one indoor unit running.
4. In two indoor units refrigerating, power consumed by compressor is larger than in only one indoor unit running.
5. In two indoor units refrigerating, EER value of the system is higher than in only one indoor unit running.

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